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by

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for

MACHINED SPRING DISPLACER FOR STIRLING CYCLE MACHINES

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FIELD

This invention relates generally to free-piston Stirling engines and more particularly to displacer construction within such engines.

BACKGROUND

A Stirling engine is characterized by having an external heat source as contrasted with an internal combustion engine. The external heat source can come from the combustion of fossil fuels, concentrated solar energy, heat from the decay of radioactive isotopes, hot exhaust gasses from diesel engines, or any other source of heat. Early Stirling engines used air, but modern ones use a gas such as Helium at high pressures to both improve performance and reduce engine physical size.

There are two main methods of transmitting forces from the Stirling power piston to perform useful mechanical work on a load such as an electrical generator. In a so-called "kinematic" design, a power piston, and a displacer piston, if utilized, are connected to a crankshaft, as in a conventional internal combustion engine. The power piston and, if applicable, the displacer piston turn a load such as a

rotary electrical generator. In this case, piston excursion is constrained to limits established by the piston's rigid mechanical connection to the crankshaft.

The second configuration is the so-called "free piston" Stirling engine ("FPSE") wherein a mechanically unconstrained power piston and displacer fundamentally move in linear simple harmonic motion at a frequency nominally equal to a natural mode determined by piston and displacer masses, various restoring spring rates provided by pneumatic, mechanical or other means, and damping effects occurring during engine operation. Typically, FPSE piston displacement is controlled by an appropriate dynamic balancing of input heat flux and mechanical loading to avoid excursions beyond design limits which would cause undesired impact with the cylinder ends. In one typical FPSE application, the power piston is connected by a rigid rod to a cylindrical magnetic structure (often called a "mover") which cooperates with the fixed portion of a linear electrical alternator. The back and forth movement of the mover/power piston generates an AC voltage at the output of the alternator.

In some applications, the FPSE configuration is preferred to its kinematic alternative, one distinct advantage being that the FPSE virtually eliminates piston-cylinder wall normal forces avoiding the need to lubricate these surfaces and means to isolate lubricant-intolerant engine components.

A cross sectional view of a generic FPSE/linear alternator (FPSE/LA) combination 10 is illustrated in FIG. 1 with the FPSE portion 50 to the left of the figure and the alternator portion 60 to the right of the figure. A gas-tight case 12 contains a freely moving displacer 14 guided by a fixed displacer rod 16. A

movable power piston 18 is connected to a permanent magnet structure 20. Various ring seals (not illustrated) may be used to form a gas tight seal between the displacer 14 and power piston 18 and internal part of the case 12. Alternatively, tight radial clearances may be used to limit leakage flows around the pistons and displacer components.

Usually, the four central spaces inside the case are denominated as follows. The space between the displacer 14 and the case 12 is the expansion space 32; the space inside the displacer 14 may serve as a gas spring 34, the space between the displacer 14 and the power piston 18 is the compression space 36; and the space between the power piston 18 and the case 12 is the bounce space 38. The case 12 may be mounted on mechanical springs (not illustrated). Thermal energy to run the Stirling engine is supplied by a heater 40 on the outside of the case 12.

Control of displacer movement both in terms of excursion and its phase relationship to the power piston motion are important factors in FPSE design. In particular, it is advantageous to configure the displacer so that it operates at or near its natural resonant frequency. By enforcing this requirement, many benefits are obtained including engine operation at or near peak efficiency (i.e. for a given input, a higher engine output is obtained).

Prior art solutions generally employ springs of various types in connection with the tuning of displacer movement to a selected resonant frequency based upon particular spring characteristics. Such springs are located within the regions 34 and 36 of the FPSE illustrated in Fig. 1. Typically, in the case of a mechanical

spring, the spring is formed as a helical wire and is linked to the displacer 14 and connected between the end of the displacer rod and its cylinder housing.

Natural resonant frequency is a function of both the mass of the collective moving body (displacer and spring) and the spring rate. A given mass-spring system can be tuned to operate at the desired frequency through the control of these two elements in conjunction with the expected damping effect during engine operation. Each particular spring has a single force constant which is determined by its material, geometrical configuration and Hooke's law.

Unfortunately, various drawbacks exist with respect to the use of springs in connection with the control of displacer movement to a particular frequency. Conventional coil springs require the use of a pair of springs deployed in opposition to one another such that the displacer can be controlled in both directions along an axial path. The need for two springs rather than one adds cost and an additional failure point. Another particular problem associated with displacer springs in FPSEs is a less than desirable component life. Prior art mechanical coil springs tend to wear out by flaking, fatiguing and ultimately failing. Various characteristics of prior art spring constructions lead to this result. For example, radially directed and side forces and/or bending moments are applied by the springs upon the displacer and the displacer rod. This can result in decreased wear life both with respect to the spring and with respect to the displacer itself. Further, these side forces increase the static friction between the walls of the displacer rod and the cylinder and can thus also have the effect of impeding initial engine starting.

Additionally, rubbing of the displacer 14 against the containing wall may result if the displacer 14 is not properly centered initially or if it moves off-center as a result of spring changes or spring movement. This is because conventionally coiled spring solutions do not provide any radial stiffness to assist in maintaining the displacer 14 centered on axis.

Another drawback associated with prior art mechanical coil spring solutions is the requirement for a pre-load wherein each of the pair of springs is under some degree of compression at all times even when the displacer 14 is in its rest position. Pre-load is needed to prevent the springs from rattling which, in turn, can cause noise and particulate contamination. Unfortunately, however, pre-loading causes higher stress levels and decreased spring life compared to what could be obtained without a pre-load. Additionally, opposed coil spring designs which are currently in use typically require the use of additional compression space and surface area within the FPSE to accommodate the spring.

Other spring configurations have also been used in connection with displacer control. For example, flat “flexure” or “planar” mechanical spring configurations have been employed in displacer control applications. While these spring configurations typically provide low wear and resulting long life, the mechanical design of the displacer must typically accommodate the unique spring characteristics resulting in more complex displacer design requirements. Additionally, “flat” mechanical spring configurations can be relatively expensive as compared to traditional coiled spring configurations.

SUMMARY

One aspect is to provide a displacer and spring assembly that addresses the drawbacks described above.

Another aspect is to provide a spring for use in connection with a displacer in an FPSE which provides an enhanced operating life in comparison to conventional mechanical coil springs.

Yet another aspect is to provide a spring for use in connection with a displacer in an FPSE which results in diminished displacer wear.

Another aspect is to provide a spring for use in connection with a displacer in an FPSE which generates minimal or no side forces or bending moments on the displacer component.

A still further aspect is to provide a spring and displacer assembly for use in connection an FPSE with reduced compression space volume and surface area requirements.

A preferred form of the displacer spring of the present invention includes various embodiments. One such embodiment calls for the use of a machined spring with multiple coils. The multiple coils of the machined spring of the present invention serve to minimize bending moments and side loads as compared to a single coil solution which results in an unbalanced load transfer to any bounding structure. Through the use of the machined spring of the present invention, the footprints of the coils may be geometrically balanced to prevent the negative

effects described above. The machined spring of the present invention operates alternately in a tension mode and a compression mode and allows the operating frequency of displacer movement to be controlled therewith to a desired resonant frequency. The machined spring of the present invention further provides enhanced structural accuracy which, in turn, leads to the minimization of lateral and side loads as compared to prior art wire-wound helical springs. Additionally, the two mechanical coiled springs typically employed in the prior art with respect to each displacer may be replaced with a single machined spring according to the teachings of the present invention.

Other embodiments of the present invention are also possible as described in further detail below and as will be understood by one of skill in the art.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention will now be described in more detail with reference to preferred forms of the invention, given only by way of example, and with reference to the accompanying drawings, in which:

FIG. 1 illustrates the basic structure of a FPSE/LA system as is known in the art;

FIG. 2 is a sectional view of the displacer/machined spring assembly according to a preferred embodiment of the present invention; and

FIG. 3 is a close-up side view of a machined spring for use in connection with displacer operating frequency control according to a preferred embodiment of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Reference is now made to the embodiments illustrated in Figs. 1-3 wherein like numerals are used to designate like parts throughout.

Figure 2 illustrates a displacer assembly 220 shown mounted to a displacer rod 260. The displacer rod 260 is attached to the engine casing (not shown). The displacer assembly 220 consists of a displacer seal body 270, a displacer cap assembly 210 and a machined spring 230. The machined spring 230 is attached to the displacer seal body 270 with mounting screws 240, and to the displacer rod 260 with a mounting screw 250.

In this way, during displacer reciprocation, one end of machined spring 230 remains fixed in place via pin assembly 240 while machined spring 230 is free to expand and contract in direct relationship with the movement of displacer 220. While it is important that machined spring 230 be free to expand and contract in direct relationship with the movement of displacer 220, the above disclosed mechanism for attaching machined spring 230 to displacer rod 260 is only one of many possible ways of providing attachment. So long as machined spring 230 is fixed in place at one and free to expand and contract at the other, the benefits of the present invention may be obtained. Thus, the invention is not necessarily limited to the disclosed embodiment for affixing machined spring 230.

In accordance with the teachings of the present invention, displacer 220 and machined spring 230 are designed such that the moving mass and the force

constant of machined spring 230 provide a combination which is mechanically resonant at the desired frequency.

According to one preferred embodiment of the present invention, machined spring 230 of the present invention may be formed according to the following specifications. Spring steel with an E value of 3.05 E7 PSI and with a P. ratio of 0.28. Such a spring may be obtained, for example, from Helical Products Company located in Santa Maria, California. According to one preferred embodiment of the present invention, the machined spring of the present invention may take the following form.

Material: 15-5PH CRES (Heat Treat H900 per AMS 5659)

Construction: Single piece machined from rod stock.

Size: ~83mm long by ~44mm outside diameter by ~28mm inside diameter.

Configuration: Two intertwined coils having ~3 turns each. The individual turns/coils are roughly 3.2mm high spanning from inside diameter to outside diameter, and are spaced roughly 3.2 mm apart. Axial positioning of the coils within the length of the spring can be varied to modify the natural frequency of the spring. Positioning the coils closer to the end of the spring that is attached to the displacer reduces the total moving mass, and thus increases the natural frequency of the system. Positioning the coils closer to the end of the spring that is attached to the displacer rod increases the total moving mass, and thus decreases the natural frequency of the system.

While the above is one preferred form of the machined spring of the present invention, it will be recognized by those of skill in the art that various other spring characteristics including varying the spring size, shape or material may be substituted for the above without departing from the scope or spirit of the present invention. For example and without limitation, the material used in forming the machined spring of the present invention is not necessarily limited to the classical "spring steels" normally used in wire coil springs – both ferrous and non-ferrous materials with the desired mechanical properties (fatigue strength and ease of manufacture) can be utilized as opposed to the case when a traditional wire coil spring is used.

Fig. 3 is a close-up side view of a machined spring that may be employed in connection with the teachings of the present invention. As can be seen, machined spring 230 includes two end portions 310 and 320 and two helical coils 330a and 330b located between end portions 310 and 320. While the embodiment in Fig. 3 shows the use of two coils, it will be understood by one of skill in the art that a larger number of coils could also be used to form the spring such as, for example, three or more coils.

Given a machined spring with the above characteristics, testing has shown that with either an axial compressive loading of 50 lbs or an axial tensile loading of 50 lbs, the maximum Von-mises stress is on the order of 30 Ksi. Further, the maximum deflection resulting from the axial load is 0.09 inches. The resulting equivalent spring stiffness for the tested spring is 554 lbf/in. Most importantly, in

the case of purely axial spring forces, testing has shown essentially no contact or wear of the springs or the displacer over approximately 200 hours of operation.

As will be apparent to one of skill in the art, the particular machined spring characteristics described above are merely exemplary and the invention may be practiced using machined springs with different physical characteristics as required or desirable in connection with various applications.

Through the use of a machined spring operating alternately in tension and compression in connection with displacer reciprocation, various benefits can be achieved. As described above, component centering can be more easily achieved as against prior art helical wire springs and rubbing can thus be easily minimized as can component wear. Further, due to the higher spring forces achievable with the machined springs used in connection with the present invention, reduced compression space volume and surface area can be achieved since the machined spring may be entirely contained within the displacer.

A machined spring and displacer/spring assembly for use in connection with an FPSE has been disclosed. It will be understood that the teachings provided above have a great many applications particularly to those associated with the control of reciprocating members in general. For example, the helical spring of the present invention may also be used in connection with other reciprocating members in Stirling engines such as with the power piston and/or in connection with the alternator. While the subject invention has been illustrated and described in detail in the drawings and foregoing description, the disclosed embodiments are illustrative and not restrictive in character. All changes and

modifications that come within the scope of the invention are desired to be protected.

What is claimed is: